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PRELIMINARY ANALYSIS OF A DOWNSIZED ADVANCED GAS-TURBINE

ENGINE IN A SUBCOMPACT CAR

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Abstract

A study was conducted to see if relative fuel economy advantages exist for a ceramic turbine engine when it is downsized for a small car. A 75 kW (100 hp) single-shaft engine currently under development was analytically downsized to 37 kW (50 hp) and analyzed with a metal-belt continuously variable transmission in a synthesized car. With gasoline, a 25-percent advantage was calculated over that of a current spark-ignition engine, scaled to the same power, using the same transmission and car. With diesel fuel, a 21-percent advantage was calculated over that of a similar diesel-engine vehicle.

Summary

A preliminary analytical study was conducted to see if relative fuel economy advantages exist for an advanced gas turbine (AGT) engine when it is downsized for a subcompact car. A particular single-shaft AGT engine, currently in development as a part of the Gas-Turbine Highway Vehicle Systems Program under Department of Energy (DOE) sponsorship, was downsized from 75 kW* (100 hp) to 37 kW (50 hp) and analyzed with a metal-belt continuously variable transmission (CVT) in a synthesized car. A vehicle inertia test mass of 964 kg (2125 lb) and a drag coefficient of 0.39 were assumed. Fuel economy sensitivity calculations were made about the set of design parameter values that were estimated for the downsized AGT engine. Comparative baseline calculations were made for current spark-ignition and diesel engines, scaled to the same power level, in the same car. The baseline calculations were made with both conventional manual transmissions and the assumed CVT. Potential differences in engine and vehicle mass were neglected and engine warm-up fuel needs were not accounted for in fuel economy comparisons. Overall dimensions of the downsized AGT engine were not estimated to see if it would fit in the subcompact car.

The downsized AGT engine was found to be a more viable option than indicated by earlier estimates of other investigators. An engine idle speed of 60 percent was needed to meet the initial acceleration distances calculated for the baseline vehicle. At this idle speed, a combined fuel economy of 29 km/l (69 mpg) was calculated for the downsized AGT engine vehicle using diesel fuel on a 15° C (59° F) day at sea level. With all engines mated to the CVT and in the same car, the downsized AGT engine using gasoline had a 25-percent better fuel economy than that of the spark-ignition engine, and using diesel fuel, had a 21 percent better fuel economy than that of the diesel engine.

Introduction

The ability to downsize automotive engines while retaining high levels of performance is important in the current marketplace. Conventional spark-ignition and diesel engines have this attribute. Reducing their size and total vehicle weight is a major means for improving fuel economy. However, there are concerns over whether advanced alternate heat engines, and in particular, the AGT engines(1,2,3) can be downsized and still retain high levels of performance and fuel economy relative to conventional engines.

DOE began the Gas-Turbine Highway Vehicle Systems Program in 1979 with an aim of demonstrating at least a 30-percent improvement in fuel efficiency over comparable vehicles powered by spark-ignition engines. The ensuing AGT projects, managed by NASA Lewis Research Center for DOE, focused on ceramic engines operating at maximum temperatures of 1371° C (2500° F), that would develop about 75 kW (100 hp) for use in 1360 kg (3000 lb) cars. Early estimates of AGT downsizing, presented by Burke and Dowdy(4), indicated that fuel economy advantages would be greatly reduced because of size-related penalties and relatively poor idle and part-load fuel consumption. At the 37 kW (50 hp) power level, Burke and Dowdy estimated a 6-percent fuel economy advantage for a single-shaft AGT over a 1985 spark-ignition engine and about a 15-percent advantage for a two-shaft AGT. A four-speed manual transmission was used for the two-shaft AGT and a split-power, hydromechanical, CVT being developed by Orshansky Transmission Corporation was used for the single-shaft AGT. Burke and Dowdy estimated that size effects would degrade specific fuel consumption from about 0.23 kg/kW-hr (0.37 lb/hp-hr) at 75 kW (100 hp) to about 0.25 kg/kW-hr (0.42 lb/hp-hr) at 37 kW (50 hp), or about 12 percent. This factor was apparently used across the operational power range of the engine in making their AGT fuel economy projections. Ronzi and Rahnke(5), in contrast, gave a recent indication of little penalty with single-shaft AGT downsizing. However, Ref. 5 gives no detail on the degree of downsizing that was considered.

This analytical study was made to re-examine AGT downsizing effects on relative fuel efficiency in light of current projections* for full-size AGT engines and in more detail than that of Ref. 4 or 5. However, the scope of the analysis was limited to one powertrain and vehicle configuration to more quickly reach a further conclusion on the viability of a downsized AGT engine. The Garrett/Ford single-shaft engine(2) was chosen for downsizing to 37 kW (50 hp) and was mated to a particular prototype, metal-belt CVT. The downsized AGT/CVT combination was analyzed in a synthesized 964 kg (2125 lb) car. Comparable baseline calculations were made for current, efficient, spark-ignition and diesel engines in the same synthesized car. The baseline calculations were made with both conventional manual transmissions and the assumed metal-belt CVT.

*English units were primary in this study.

*The projected design specific fuel consumption for the full-size AGT engine of Ref. 2 is about 0.18 kg/kW-hr (0.30 lb/hp-hr).

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The results here are further qualified in that no downsized AGT layout was made, and differences in engine weight and warm-up fuel needs were not considered in the comparisons. Effects of warm-up heat losses on fuel economy are presented for the downsized AGT engine/vehicle.

Overall results of this study were presented at the Automotive Technology Development Contractor Coordination Meeting in Dearborn, Michigan, on October 27, 1981. This report presents more details of the study, including results of a fuel economy sensitivity analysis for the downsized AGT vehicle and the effect of variable compressor inlet guide vanes (VIGV) on fuel economy which were not previously presented. Details of the approach are presented first under METHODS, followed by results and discussion. Symbols are defined in Appendix A.

Methods

This study required downsizing loss estimates for the AGT single-shaft engine, performance map calculations for downsized AGT engines, and fuel economy and acceleration performance calculations for both the down-sized AGT and conventional engines in a subcompact car. Two computer codes were used: one for gas-turbine performance; the other, for both vehicle fuel economy over the Federal combined driving cycle and wide-open-throttle vehicle acceleration performance. Baseline spark-ignition and diesel vehicles were synthesized for analysis in the vehicle code. A 1981 version of the Dodge Colt and a 1980 model of the Volkswagen Diesel Rabbit were used. Computed fuel economies were compared to published EPA measurements for these vehicles to check overall accuracy of the modeling. For relative comparisons, all engines were scaled to 37 kW (50 hp) and analyzed in a car at the inertia test mass of the Dodge Colt (964 kg or 2125 lb). The baseline spark-ignition and diesel engine maps were scaled linearly based on the ratios of rated power while maintaining measured specific fuel consumptions. No attempt was made to account for differences in engine weight. The car simulated for these relative comparisons was also assumed to have reduced aerodynamic drag to better represent future vehicles. The downsized AGT vehicle was forced to meet the initial acceleration distances calculated for the downsized spark-ignition vehicle. This was accomplished through a selection of AGT engine idle speed. All vehicles in the calculations were screened to achieve at least 97 kph (60 mph) on a 4-percent grade.

A sensitivity analysis was made to examine the importance of parameter values that were selected or estimated for the downsized AGT engine. Each parameter was varied around the base values, one at a time. With each parameter change, the engine was re-designed to achieve its rated 37 kW (50 hp) and analyzed for its performance map. Fuel economy was also re-calculated.

Further assumptions, limitations, and general procedures are discussed under the sub-headings: Computer Codes, AGT Engine Scaling, and Vehicle Synthesis.

Computer Codes

The gas-turbine performance code used in this study was a modified version of the Navy/NASA Engine Program (NNEP)[6]. The modifications to NNEP are described in Ref. 7. Subroutines in NNEP allow three-dimensional interpolation through curve fits, and dependent parameter optimization. These features allowed use of compressor maps with VIGV and the optimization of VIGV settings for least fuel flow. The same lower heating value of 42,800 J/g (18,400 Btu/lb) was used for both gasoline and diesel fuel to calculate required fuel-flow rates.

The vehicle performance code includes subroutines for fuel economy and acceleration calculations. It is an undocumented internal Lewis code. However, fuel economy calculation methods are described in Ref. 7. The code uses steady-state engine performance maps and no correction is made for cold-start fuel penalties associated with the city driving cycle in the Federal Test Procedure. Fuel economies were calculated assuming a density of 0.739 kg/l (6.17 lb/gal) for gasoline, and 0.849 kg/l (7.09 lb/gal) for diesel fuel. All powertrain inertia effects were neglected in fuel economy calculations and dynamometer procedures were used to determine power requirements (see Appendix B). The vehicle acceleration calculations were based on the inertial effects of the vehicle, engine, and wheel assemblies and also included vehicle weight shifts between axles and tire traction limits. Transmission inertia was assumed to be small and was neglected. Transient temperature lags were not calculated.

AGT Engine Scaling

The basis for downsizing was the original design point for the full-size engine which resulted in 97 kW (130 hp) at 100,000 rpm on a 29° C (85° F) day at sea level. The full-size AGT engine is currently flat-rated, through speed controls, to provide a peak power of 75 kW (100 hp) over a wide range of ambient temperatures and pressures. Major design parameters were maintained with downsizing; namely, the design-point ambient temperature and pressure, the peak turbine-inlet and regenerator-inlet temperature, the peak turbine-tip speed, and the design-point compressor pressure ratio and specific speed. Selection of these same parameter values at the 37 kW (50 hp) power rating resulted in an increase in maximum shaft speed and a decrease in turbomachinery-tip diameters.

Initially, sizing subroutines in the modified version of NNEP were used to estimate design-point compressor and turbine efficiencies and diameters for both engine sizes. Resulting efficiencies were further corrected to maintain constant impeller tip clearances between the engines. Turbine-tip-clearance correlations were from Ref. 8. Similar compressor-tip-clearance correlations were from unpublished Lewis data. Net calculated differences in efficiency between the 97 kW and the downsized 37 kW cases were then applied to the AGT contractor's estimates for the full-size compressor and turbine efficiencies to obtain the estimated efficiencies for the downsized AGT engine.

The regenerator was assumed to be geometrically similar for both engines. Its effectiveness was unchanged on the assumption that the flow per unit of cross-sectional area could be maintained by changes in disk diameter about the same mean diameter using the same ceramic matrix and disk thickness. Regenerator seal leakage, expressed as a percentage of engine airflow, was assumed a function of seal length and mass-flow rate. This assumption resulted in the percent of regenerator seal leakage varying inversely proportional to the

square root of airflow. All other flow leaks were scaled assuming the same gap clearances at all leakage points, resulting in the leakage areas varying with the ratio of compressor-tip diameters. Heat leaks were assumed to vary at the same percentage as the flow leaks. This assumption is believed to be reasonable for energy transfer between components, but perhaps overestimates heat losses from the engine. From the sensitivity results to be presented later, the reader will find that fuel economies are insensitive to small changes in either heat or flow losses. Relative ducting pressure drops were assumed to be the same between engines.

An estimate of shaft diameter for the downsized engine was made by maintaining the same shaft shear stress as the reference engine. Shaft size influences bearing size and power loss. Bearing parasitic losses were then scaled proportional to rotational speed squared and shaft diameter cubed.

All off-design losses in the downsized engine were assumed to vary proportional to those in the full-size engine. That is, the off-design component performance maps in NNEP were the same as those projected analytically for the full-size AGT engine, but normalized to the design parameter values for the downsized engine.

Vehicle Synthesis

The synthesis of a vehicle for computer simulations requires the assembly of component performance data and specific vehicle characteristics. Often, as was the case in this study, not all of the required data can be obtained for a specific vehicle. Some proprietary component data was used, and some missing data were substituted with similar data that was available. Sources for component data are presented under Powertrain which follows the discussion of vehicle characteristics.

Vehicle Characteristics. Vehicle parameter values are presented in Table I. For both baseline vehicles, fuel-economy test mass was set at 45 kg (100 lb) over the fully-fueled curb mass to simulate one passenger* and 40 percent fuel mass as prescribed for the Environmental Protection Agency (EPA) test procedures. Performance test mass was assumed to be 91 kg (200 lb) over the economy test mass to simulate two passengers* and a full fuel tank. Inertia masses and PAU settings are those published by the EPA with fuel economy and emission measurements for the baseline vehicles. The vehicle with improved aerodynamics was assumed to have the same masses as the Dodge Colt. The improved aerodynamic assumptions, presented in Appendix B, result in a vehicle drag coefficient of 0.39. This drag coefficient is a little better than that obtained by General Motors' X-body cars.

The remaining parameters in Table I were common for all vehicles. The assumed rolling resistance coefficient is approximately that of current steel belted radial-ply tires. The only vehicle accessory was assumed to be an alternator with the power needs shown in Fig. 1.

Powertrain: Both baseline engine maps were obtained from the Transportation Systems Center of the Department of Transportation (DOT) at Cambridge, Massachusetts. The Colt map was provided by Mitsubishi Motors with a purchase of this engine for testing at DOT and was proprietary. The diesel Rabbit map was a result of DOT testing which has not been published. The metal-belt CVT data used here was also proprietary and was the result of testing done at Borg Warner Corporation in Des Plaines, Illinois. This particular CVT was a VanDoorne Transmissie BV prototype designed for use in a European version of the Fiat Strada called the Ritmo.

Table II presents values for powertrain parameters. The CVT had gear ratios ranging from 2.3 to 0.59 for an overall ratio range of 3.9. The final drive ratio for this CVT was 6.24, yielding overall transmission ratios from 14.4 to 3.7. The form of the test data is shown in Fig. 2 for the highest belt ratio. At each input torque level, the efficiency test data was assumed to be constant down to an input speed of 750 rpm. To simulate a slipping clutch for start-ups, the data for the highest belt ratio was further extrapolated to zero efficiency at zero speed. The CVT oil pump and final drive losses were included in the test data. Two forward drive gears were assumed for use with the AGT engine to provide an overall ratio range of about 15. Reduction gearing was also needed between the AGT output shaft and the clutch input shaft. An additional 4-percent power loss was assumed for the extra gearing on the basis of a 1-percent loss for each gear set and allowance for two gear sets in the reduction gearing.

The particular Colt vehicle chosen for simulation had a 4-speed manual transmission and a final drive ratio of 3.47. The chosen Rabbit vehicle had a 5-speed manual transmission and a final drive ratio of 3.90. Individual gear ratios are shown in Table II. Mechanical gear efficiencies and output spin losses for the 4-speed manual transmission were those from Ref. 9 for the THM-125 transmission of General Motors. The corresponding values for the 5-speed transmission were extrapolations of the THM-125 data. A drive-axle efficiency of 0.97 was used for both baseline vehicles. Bookkeeping of AGT engine accessory losses was accounted for in the vehicle computer code and is reflected in the higher idle power requirement for the AGT engine. Typical shift schedules were used for the manual transmissions in fuel economy calculations. Figure 3 shows the schedules used with the four-speed transmission.

Overall transmission efficiencies were calculated in the vehicle code and time averaged for those periods when the transmission was transmitting power. A value of transmission efficiency for the combined driving cycle was obtained by harmonic averaging the constituent values in the same manner as in the fuel economy calculation.

Results

Scope of this analysis was limited to a downsizing of one particular AGT engine and its resulting performance when mated to a prototype metal-belt CVT in a subcompact car. For relative comparisons, baseline calculations were made for spark-ignition and diesel engines, scaled to the power rating (37 kW or 50 hp) of the downsized AGT, in the same vehicle and at the same mass. Baseline calculations were made with both manual transmissions and the assumed CVT.

*The mathematics imply that each passenger has a mass of about 68 kg (150 lb).

Initial study selections of a 37 kW power rating in about a 950 kilogram car resulted in somewhat underpowered vehicles by today's standards. This power-to-mass selection was made to favor fuel economy at the sacrifice of some acceleration performance. Whether this would be acceptable to a future consumer is unknown.

Baseline vehicle results are presented first to establish modeling accuracy, relative levels of fuel economy, and acceleration requirements for the AGT vehicle. Downsized AGT engine and vehicle results are presented next, followed by vehicle comparisons among engine/transmission types. Results of the AGT vehicle sensitivity analysis are presented last.

Baseline Vehicles

Fuel economy results for the baseline vehicles are presented in Table III. Part A shows results for the spark-ignition engine and, part B, those for the diesel engine. The first two data columns compare results between EPA measurements and the base vehicle simulation cases. The calculated highway fuel economies were lower than the measured values, while the city economies agreed. If a cold-start correction (about -3 percent based on DOT testing on this engine-size class) were applied to the calculated city fuel economies, they too would underestimate the measured values. The simulated combined fuel economies were about 2 percent lower (4 to 5 percent, with the estimated cold-start penalty) than the measured values. In any case, the authors felt the agreement to be close enough to validate the overall modeling.

The remaining columns in Table III show the separate added effects on calculated fuel economies of reduced drag, mass change (diesel, part B only), engine power scaling, and changes from the manual transmissions to the assumed CVT. Effects of reduced vehicle drag were greater on the highway values as would be expected from the higher speeds during this cycle, and were also greater on both highway and city values for the heavier diesel vehicle. The mass reduction then increased each fuel economy for the diesel vehicle by about 1 km/l (2 mpg). The spark-ignition engine was scaled down to 37 kW (50 hp) and gained about 1 km/l (3 mpg), while the diesel was scaled up slightly and lost less than 1 km/l (or about 1 mpg).

Effects of the transmission change on combined fuel economy, resulted in less than a 7-percent gain for the spark-ignition engine and about a 12-percent gain for the diesel engine. Comparisons in Table III show that this was due to the relatively small increase in city fuel economy with the spark-ignition engine compared to that with the diesel engine. Inspection of the spark-ignition map showed elongated islands of constant specific fuel consumption with output speed in comparison to those of the diesel-engine map and other internal-combustion-engine maps. And these islands, particularly those near minimum specific fuel consumption, did extend to low output speeds. Therefore, it appears that the baseline spark-ignition engine has been tailored to city operation and does not benefit as much as the diesel engine from operation with the CVT.

Lower drive-axle ratio models of the base vehicles were produced and had somewhat higher fuel economies. However, upon scaling to the reference vehicle conditions, these options were calculated to be unable to meet the imposed 4-percent grade criterion.

Wide-open-throttle acceleration results for the baseline vehicles are presented in Table IV. Acceleration times, from 0 to 97 kph (0 to 60 mph) and 2- and 4-second distances are shown. The general format is similar to that in Table III, except that there is no comparison to measured values. Test data for these base vehicle models were not available; however, the calculated acceleration times appear to be representative of similar vehicles. All of the calculated values in this table were based on the normal engine idle speeds; that is, higher kick-off speeds that can be developed with manual transmission were not used. Typical calculated effects of kick-off speed are shown in Fig. 4 for the spark-ignition vehicle.

Effects of power scaling on acceleration of the spark-ignition vehicle, Table IV, added 3.5 seconds to the 0-to-97 kph time and decreased initial distances. The combined effect of mass reduction and power scale-up on the diesel vehicle decreased the required acceleration time by 4.4 seconds and increased initial distances. Use of the CVT in both vehicles resulted in substantial acceleration improvements. For example, acceleration time for the spark-ignition vehicle was reduced by 2.9 seconds and for the diesel vehicle by 3.7 seconds. Comparison of results showed that the CVT allowed sufficient torque at the vehicle drive wheels to cause tire-slip-limited accelerations of 3.7 m/s^2 (12.3 ft/sec^2) up to vehicle speeds of about 27 kph (17 mph), while this peak acceleration occurred only momentarily in first gear with the manual transmissions.

Based on the results in Table IV, the authors decided to use the initial acceleration distances calculated for the spark-ignition vehicle with the four-speed transmission as criteria for suitable response of the downsized AGT vehicle with the CVT; that is, 5.0 m (16 ft) in the first 2 seconds and 20 m (67 ft) in 4 seconds.

AGT Engine and Vehicle

Table V presents results of AGT engine downsizing estimates on design-point parameters and a comparison to those for the original design point. The full-size engine is currently flat-rated at 75 kW (100 hp). Maintaining major thermodynamic parameters required an increase in peak shaft speed from 100,000 to 156,000 rpm. Turbomachinery tip diameters were estimated to be reduced to about 6.9 cm (2.7 in) for the radial compressor and about 8.6 cm (3.4 in) for the radial turbine. The turbomachinery efficiencies were estimated to be reduced by 0.013 for the compressor and by 0.018 for the turbine.* Regenerator seal leakage fraction was

*These turbomachinery efficiency decrements may appear small to the reader based on individual turbine and compressor size correlations in the literature. It is important to note that the results here are based on Lewis correlations that were iterated on in NNEP and result in new geometry for both the turbine and compressor. The reader is also directed to the sensitivity results presented later which allow the calculation of effects of larger turbomachinery efficiency decrements on fuel economy.

increased by 54 percent, while shaft bearing and seal losses were reduced by 43 percent. Resulting inlet mass-flow rate was reduced by 58 percent.

Figure 5 shows a comparison of specific fuel consumption between the AGT engines on a 15° C (59° F) day. Compressor VIGV settings were optimized to provide least fuel-flow rate at each power output. At rated powers, the smaller AGT had a specific fuel consumption that was 0.02 kg/kW-hr (0.03 lb/hp-hr) greater than that of the larger engine. Comparison of minimum specific fuel consumptions shows the downsized engine to be about 0.01 kg/kW-hr (0.02 lb/hp-hr) higher. The curves cross at about 20 kW (27 hp), with lower specific fuel consumptions for the downsized engine below that power level. This was particularly important since nearly all engine power needs for the small car over both city and highway cycles occurred below 20 kW. The highest power need occurred in the city cycle during the highest acceleration and was about 25 kW (34 hp). Steady cruising at 97 kph (60 mph), for example, required only 13 kW (17 hp).

Effects of idle speed on fuel economy and acceleration performance of the downsized AGT engine are shown in Fig. 6. As with any gas-turbine engine, vehicle response improves with increasing idle speed while fuel economy decreases. The baseline vehicle response criteria are indicated by the circles in Fig. 6. To meet both the 2- and 4-second distances, an AGT engine idle speed of 60 percent was required. At 60-percent idle, combined fuel economy was 29 km/l (69 mpg) using diesel fuel. Acceleration time was 17.0 seconds.

A dual engine idle speed approach has been proposed for the full-size engine and transmission combination[2]. In this concept, the AGT engine would idle at about 50 percent during stops and brake-on operation. The second or higher engine idle speed would be reached through automatic control system actions in the time it takes the driver to move his foot from the brake to the accelerator pedal. Such an approach, if practical, would provide both responsiveness and higher fuel economy. If such a delayed speed start-up clutch arrangement was found to be practical for the downsized AGT/CVT combination, a combined fuel economy of 31 km/l (73 mpg) would be indicated from the results in Fig. 6 at a 50-percent lower idle speed.

The importance of compressor VIGV to engine and vehicle operation is shown in Fig. 7 and Table VI. Figure 7 shows the effect on specific fuel consumption and Table VI presents the resulting effect on fuel economy. Although the change in engine specific fuel consumption in Fig. 7 appears to be small, the relative change increased rapidly with decreasing power. And, at idle conditions, fuel consumption decreased by 34 percent with VIGV operation. Without VIGV, low power outputs were reached by reductions in turbine-inlet temperature below those which would be limiting for the regenerator. With VIGV, power outputs at any engine speed could be reached with operation at limiting temperatures. That is, the VIGV provided sufficient mass-flow control to allow power reduction without further decreases in fuel-flow rate. The effect of no VIGV on combined fuel economy (Table VI) was a reduction of 3 km/l (7 mpg), or 10 percent. The effect of VIGV operation on low-power operation is also reflected in the larger change in city fuel economy as compared to that on the highway.

Vehicle Comparisons

A comparison of fuel economies among engine types is presented in Table VII. Each engine was scaled to a rated 37 kW (50 hp) and analyzed with the assumed CVT. All vehicles had the same mass and reduced drag. The spark-ignition vehicle is compared to the downsized AGT vehicle using gasoline, while the diesel vehicle comparison is made using diesel fuel. On the basis of combined fuel economies and for equivalent vehicle response, the AGT engine was 25 percent better than the spark-ignition engine and 21 percent better than the diesel engine. Similar comparisons are shown in Table VIII among engine/transmission types. The AGT/CVT combination showed a 33-percent better combined fuel economy than that of the spark-ignition/manual transmission and a 35-percent advantage over that of the diesel/manual transmission. Thus, it appears that with downsizing and improved low-power fuel consumption relative to the full-size engine, the AGT still exhibits sizable fuel economy advantages over conventional engine/transmission combinations at the 37 kW (50 hp) power rating.

Table IX presents a summary of wide-open-throttle accelerations among the engine and transmission combinations. With the response criteria matched, the AGT vehicle, Table IX-A, was 2.4 seconds faster in acceleration time than the baseline spark-ignition vehicle, with its manual transmission, and 3.0 seconds faster than the baseline diesel vehicle. However, comparisons in part B with the CVT showed the AGT response to be somewhat poorer than that of the baseline vehicles. This was because of a higher engine inertia for the AGT compared to the baseline engines. A tire-slip-limited acceleration was obtained with the AGT/CVT, but it was delayed to a vehicle speed near 27 kph (17 mph).

Comparisons of transmission efficiencies are made in Fig. 8 and Table X. Figure 8 shows results for constant vehicle speed, while Table X shows values averaged over the driving cycles. Results between transmission types for the same baseline engine showed about the same efficiencies. Comparison between the AGT/CVT and the baseline engine/CVT combinations, however, showed somewhat lower efficiencies with the AGT engine. Inspection of results showed that AGT operation with this assumed CVT suffered because of lower input torques than those provided by the baseline engine. The lower input torques resulted in lower efficiencies (see Fig. 2). A metal-belt CVT designed for AGT operation with better speed/torque matching might offer further transmission efficiency gains and, therefore, still better AGT fuel economy.

AGT Vehicle Fuel Economy Sensitivity

The reference AGT engine at the original 97 kW (130 hp) rating was designed for a compressor pressure ratio of 5 and a compressor specific speed of 0.753. For the scaled 37 kW (50 hp) downsized engine, these were maintained at the same value. But pressure ratios from 4 to 5 were considered to determine the effect on vehicle performance. The sensitivity to rated pressure ratio was investigated for a constant compressor specific speed case, and for a constant shaft speed case. The results for both methods are shown in Fig. 9. Combined fuel economy is shown as a function of pressure ratio with the constant specific speed case plotted

as a solid line and the constant rotational speed case as a dashed line. The rotational speed and compressor specific speed using either method are also shown in Fig. 9. For a constant compressor specific speed, the rotational speed increased as pressure ratio increased. For constant shaft rotational speed, the specific speed decreased as pressure ratio increased. The important fact to note is that the choice of 5 for compressor design pressure ratio is a good choice using either criteria. The pressure ratio of 5 may not be the exact optimum but it is close to optimum when looking at combined fuel economy.

The compressor specific speed of 0.753 was kept the same as the Ref. 97 kW (130 hp) AGT engine. Compressor specific speed at the design point was varied over a range to determine the effect on vehicle fuel economy. The compressor design pressure ratio was held constant at 5. The result is shown in Fig. 10 where vehicle combined fuel economy and shaft rotational speed are shown plotted against compressor specific speed. The reference point specific speed of 0.753 is shown. While there is no clear best choice for compressor specific speed, lowering specific speed lowered rotational speed but also lowered combined fuel economy. If rotational speed were limited by some particular constraint then that would influence the choice of compressor specific speed. Dropping specific speed from .75 to .5 would lower the combined fuel economy from 29.5 km/l (69.3 mpg) to 27.9 km/l (65.6 mpg).

The sensitivity of the combined fuel economy to several other parameters will now be examined. In addition to the vehicle inertia mass two categories of parameters will be considered for their impact on vehicle fuel economy. The impact of lower than expected component efficiencies is one category of parameters to be considered. The other category involves those losses that the designer attempts to minimize but cannot be completely eliminated. Flow leaks, heat leaks, bearing parasitic losses, and the quantity of warm-up fuel fall into this category.

Table XI is a list of most parameters which were varied to obtain the sensitivity of their impact on vehicle fuel economy. The range over which each parameter was varied is shown as well as sensitivity coefficients in two forms. The second last column lists the influence coefficient which represents the percent change in fuel economy for a 1-percent change in the value of the parameter. The slope at the reference value of a plot of vehicle combined fuel economy against the parameter being varied is tabulated in the last column. The individual plots are not presented.

Vehicle inertia mass was varied from 873 to 1055 kg (1925 to 2325 lb) with 964 kg (2125 lbm) being the reference value. For each 1-percent increase in vehicle mass combined fuel economy decreased 0.37 percent. This amounts to -0.01 km/l for each kilogram increase in mass.

Turning to the category of component efficiencies next the efficiency for reduction gearing required between the AGT output shaft and clutch input shaft was assumed to be 96 percent efficient. Fuel economy sensitivity to that gearing efficiency was obtained by varying gear efficiency from 90 to 100 percent efficiency. The combined fuel economy decreased by 0.66 percent for each 1-percent decrease in gearing efficiency. The component efficiency having the greatest impact on the combined fuel economy was the regenerator effectiveness. The estimated effectiveness at rated full power was 0.929 (at part load the mass flow is lower and effectiveness is higher). If the rated full power regenerator effectiveness was lowered by 0.01 to 0.919, the combined fuel economy would be lower by 0.45 km/l (1.07 mpg). The calculated influence coefficient shows a 1.49-percent decrease in combined fuel economy for each 1-percent decrease in regenerator effectiveness.

The impact of turbine efficiency on vehicle fuel economy was the next largest in magnitude. For a 0.01 reduction of turbine total efficiency the combined fuel economy dropped by 0.32 km/l (0.76 mpg). The calculated influence coefficient for fuel economy shows a 1.08 percent decrease for each 1-percent decrease in turbine efficiency. The next most important component efficiency is that of the compressor. A reduction of design point compressor efficiency by 0.01 would lower the combined fuel economy by 0.26 km/l (0.62 mpg) which represents a 0.76 percent decrease in fuel economy for each 1-percent reduction in efficiency.

Certain losses were examined to see how they impacted vehicle combined fuel economy. The flow leakage across the regenerator face seal was estimated to be 6.3 percent of the total mass flow. Regenerator seal leakages from 2 to 8 percent were considered when examining sensitivity. If the regenerator leakage was increased from 6.3 to 7.3 percent of flow, the combined fuel economy would fall by 0.22 km/l (about 0.52 mpg). Thus, for each 1-percent increase in the leakage fraction, the vehicle combined fuel economy was reduced by 0.05 percent. Flow leaks other than regenerator face seal were lumped into one group. Each leak was individually increased over a range from 1 to 2 times the flow leaks projected for the full-size engine. The combined fuel economy declined 0.03 percent for each 1-percent increase in flow leakage. The flow leak that most affected the combined fuel economy was the regenerator face seal leakage.

In a similar fashion the effect of heat losses upon fuel economy was examined by multiplying each heat loss by a factor from 1 to 2 times the reference values. The heat leak changes were handled collectively. Some of the heat losses were from the engine to ambient and one leak represented heat transfer from the turbine hot section back to the compressor section. When heat leaks increased by 1 percent combined fuel economy decreased by 0.03 percent.

The impact of the engine idle speed on fuel economy was found by varying the idle horsepower. For the reference downsized AGT engine the best estimate was that idle power would be about 0.79 kW (1.06 hp) with idle speed 60 percent of rated speed. To obtain the fuel economy sensitivity, idle power was varied from 0 to 2.2 kW (0 to 3 hp). For a 1-percent increase in idle horsepower vehicle combined fuel economy was lower by 0.03 percent. For a 1 kW increase of idle power, combined fuel economy would be reduced by 1 km/l.

Shaft losses, which include bearing losses, were estimated at 1.2 kW (1.6 hp) for the downsized AGT engine. The bearing losses were a function of

speed. For the given reference design the shaft loss was 1.2 kW at 100 percent speed and 0.73 kW (0.98 hp) at the 60 percent idle speed condition. To account for a bearing design change or underestimation of losses, a range of parasitic power losses from 0 to 2.2 kW (0 to 3 hp) at the rated conditions was examined. At less than design rated rotational speed the losses were scaled in the same manner as the reference case. For a 1-percent increase in the assumed parasitic shaft losses the combined fuel economy was reduced by 0.11 percent. If the parasitic power increased by 1 kW, the decrease in vehicle combined fuel economy would be 2.6 km/l.

The sensitivity of combined fuel economy to engine warm-up heat loss is shown in Fig. 11 for the downsized AGT engine in the 964-kg (2125-lb) vehicle. As a point of reference, the contractor for the two-shaft AGT(1) has made a preliminary estimate of about 6.5×10^6 J (6100 Btu) in heat loss for their full-size engine. The warm-up penalty for the downsized AGT engine herein should be somewhat less. An approximation for the downsized AGT might be about 2/3 that of the full-size engine, or about 4.3×10^6 J (4000 Btu), which from Fig. 11 would reduce combined fuel economy by 6 to 7 percent. Figure 11 covers a wide range of heat loss and provides a means to estimate the impact of a more refined estimate of warm-up penalties.

Concluding Remarks

A natural question, which was not answered by this study, is how does the relative fuel economy advantage of the downsized AGT engine compare to the relative fuel economy advantage of the full-size AGT engine. Although the goal of the full-size AGT engine program was a 30-percent improvement in fuel efficiency, comparison to the results here cannot be made because of differences in transmissions and vehicles. A general result, however, was that a sizeable advantage was found for a downsized AGT engine/vehicle compared to two equivalent, but not necessarily the best, conventional engine/vehicles. A broader comparison, such as by Ronzi and Rahnke(5), including other advanced powertrain options, would have been desirable.

It is also important to iterate that several factors have been omitted in the fuel economy comparisons. The omitted factors, which could increase the fuel economy advantages of the downsized AGT, include better AGT/CVT matching and the use of dual idle speeds. On the negative side, the most important omitted factors are relative engine warm-up penalties and potential engine size or volume constraints. If the approximation for AGT warm-up penalties on fuel economy were correct (-6 to -7 percent) and dual idle speeds were practical (+6 percent), the combined effects would nearly balance. However, if the regenerator effectiveness were also reduced by size constraints, the sensitivity results would indicate about a 1.5-percent loss in fuel economy for each 1-percent loss in effectiveness. On this basis, the authors believe that the fuel economy advantages for the downsized AGT, indicated here, may be optimistic by a few percent, but would still be high enough to offer AGT downsizing as a viable option for future development efforts.

Summary of Results

A preliminary analytical study was conducted to see if relative fuel economy advantages exist for an advanced gas turbine (AGT) engine when it is downsized for a subcompact car. Initial study selections limited the scope of the study to a particular combination of a single-shaft AGT engine and metal-belt continuously variable transmission (CVT). The AGT engine was downsized to 37 kW (50 hp) and analyzed in a synthesized car with an inertia test mass of 964 kg (2125 lb) and an assumed drag coefficient of 0.39. Comparative baseline calculations were made for current spark-ignition and diesel engines, scaled to the same power level, in the same car at the same assumed masses. Engine warm-up fuel needs were not accounted for in fuel economy calculations. And overall dimensions of the downsized AGT engine were not estimated to see if it would fit in the assumed vehicle. Results are summarized as follows:

(a) The downsized AGT engine/vehicle with a 60 percent engine idle speed and using the CVT met the initial acceleration distances of the baseline vehicles with manual transmissions and was calculated to have a combined fuel economy of 29 km/l (69 mpg) using diesel fuel on a 15°-C (59°-F) day at sea level.

(b) With all engines mated to the CVT, and in the same car, the downsized AGT engine using gasoline had a 25-percent better fuel economy than that of the spark-ignition engine, and using diesel fuel had a 21-percent better fuel economy than that of the diesel engine.

(c) Compared to baseline calculations with manual transmissions, the downsized AGT/CVT using gasoline had a 33-percent better fuel economy than that of the spark-ignition engine, and using diesel fuel had a 35-percent better fuel economy than that of the diesel engine.

(d) If a dual idle speed proves practical for the downsized AGT engine, allowing a brake-on idle speed of 50 percent, the AGT vehicle could still meet the initial acceleration distances of the baseline vehicles and would have a combined fuel economy of 31 km/l (73 mpg) using diesel fuel.

(e) Combined fuel economy for the downsized AGT vehicle was reduced by 10 percent, or 3 km/l (7 mpg), when analyzed without use of variable compressor-inlet guide vanes.

(f) Sensitivity results showed that a design compressor pressure ratio of 5 was still near the best value for peak fuel economy with downsizing of the AGT engine.

Appendix A

Symbols

A_F vehicle frontal area, m^2 (ft^2)
C_A aerodynamic resistance coefficient, $\frac{N\text{-}hr^2}{m^2\text{-}km^2}\left(\frac{lbf\text{-}hr^2}{ft^2\text{-}mi^2}\right)$
C_R rolling resistance coefficient, N/kg (lbf/lbm)
f tire-to-rolls coefficient
F force, N (lbf)
K conversion constant, 3600 s/hr $\left(375\ \frac{mi\text{-}lbf}{hp\text{-}hr}\right)$
n dynamometer friction exponent
P power, kW (hp)
V vehicle speed, kph (mph)

W_E economy test mass, kg (lbm)
W_P performance test mass, kg (lbm)
X fraction of vehicle mass on drive axle

Subscripts

B dynamometer brake
D drag
DY dynamometer
M match
PAU power absorber unit
R rolling resistance
RR reference rolling resistance
T tractive

Appendix B

Road-Load and Dynamometer Equations

One aim of the Federal Test Procedures is to produce consistent emission and fuel economy test results among the various makes and models of automotive vehicles. The procedures require dynamometer testing which approximates actual vehicle performance over prescribed city and highway driving cycles. Flywheels, or other equivalent means, are used in the dynamometer tests to simulate vehicle inertia, and the Power Absorption Unit (PAU) on the dynamometer is adjusted to match "road-load" power of the vehicle at 80.5 kph (50 mph). Road-load power is that power supplied by the engine at the drive wheels on a level road under balanced wind conditions and at a constant vehicle speed. Dynamometer characteristics result in a slightly different variation with vehicle speed in the power required at the vehicle drive wheels than that from road-load tests or equations.

Road-load and dynamometer equations are presented below and are evaluated for the constants assumed in this analysis. The road-load equations were used for vehicle acceleration calculations, while the dynamometer equations were used for fuel economy calculations.

Road-load equations. Additional parameter values assumed here included:

$$A_F = 1.64\ m^2 (= 17.7\ ft^2)$$

$$C_A = 0.0185\ \frac{N\text{-}hr^2}{m^2\text{-}km^2} \left(= 0.001\ \frac{lbf\text{-}hr^2}{ft^2\text{-}mi^2} \right)$$

With no wind on a level road, and at constant vehicle speed, the tractive force at the drive wheels is given by

$$F_T = F_R + F_D \tag{B1}$$

The rolling resistance force may be expressed as

$$F_R = C_R W_P \tag{B2}$$

or,

$$\left. \begin{aligned} &F_R = 0.0981x987 = 96.8,\ N \\ &(F_R = 0.01x2176 = 21.8,\ lbf) \end{aligned} \right\} \tag{B2a}$$

The vehicle drag force may be expressed as

$$F_D = C_A A_F V^2 \tag{B3}$$

or,

$$\left. \begin{aligned} &F_D = 0.0185x1.64\ V^2 = 0.0303\ V^2 \\ &(F_D = 0.001x17.7\ V^2 = 0.0177\ V^2) \end{aligned} \right\} \tag{B3a}$$

And, for V_M = 80.5 kph (= 50 mph),

$$F_D = 0.0303(80.5)^2 = 196,\ N$$

$$(F_D = 0.0177(50)^2 = 44.3,\ lbf)$$

Since power is related to force by

$$P = \frac{FV}{K}, \tag{B4}$$

the tractive power at the vehicle wheels is

$$P_T = \left(\frac{C_R W_P}{K}\right) V + \left(\frac{C_A A_F}{K}\right) V^3 \tag{B5}$$

or,

$$\left. \begin{aligned} P_T &= \left(\frac{96.8}{3600}\right) V \left(\frac{0.0303}{3600}\right) V^3 \\ &= 0.0269\ V + 8.42\ V^3 \times 10^{-6} \\ \Bigg(P_T &= \left(\frac{21.8}{375}\right) V + \left(\frac{0.0077}{375}\right) V^3 \\ &= 0.0581\ V + 4.72\ V^3 \times 10^{-5} \Bigg) \end{aligned} \right\} \tag{B5a}$$

And, for V_M = 80.5 kph (= 50 mph),

$$P_T = 0.0269(80.5) + 8.42(80.5)^3 \times 10^{-6}$$

$$= 2.17 + 4.39 = 6.56\ kW$$

$$\left(\begin{aligned} P_T &= 0.0581(50) + 4.72(50)^3 \times 10^{-5} \\ &= 2.91 + 5.90 = 8.81,\ hp \end{aligned} \right)$$

Equation (B5a) is plotted in Fig. 11.

Dynamometer equations. A typical clayton dynamometer was assumed with the following characteristics:

$$f = 2.0$$

$$n = 1.7$$

$$P_B = 2.92, \text{ kW } (3.91, \text{ hp})$$

Vehicle drive wheels are placed on the rolls such that the rolling resistance power absorbed by the dynamometer is given by

$$P_{DY,R} = \frac{fC_R X W_E V}{K} \quad \text{(B6)}$$

or, for V_M,

$$P_{DY,RR} = \frac{2.0 \times 0.0981 \times 0.61 \times 896 \times 80.5}{3600} = 2.44, \text{ kW}$$

$$\left(P_{DY,RR} = \frac{2.0 \times 0.01 \times 0.62 \times 1976 \times 50}{375} = 3.27, \text{ hp}\right)$$

To match the road-load requirement, the dynamometer PAU setting is given by the difference between equations (B5) and (B6) evaluated at V_M, or

$$P_{PAU} = 6.56 - 2.44 = 4.12, \text{ kW}$$

$$(P_{PAU} = 8.81 - 3.27 = 5.54, \text{ hp})$$

The PAU power is then the sum of required brake power and dynamometer friction. And total steady-state power absorbed by the dynamometer is given by

$$P_{DY} = P_B(V/V_M)^3 + (P_{PAU} - P_B)(V/V_M)^n + P_{DY,RR}(V/V_M) \quad \text{(B7)}$$

And for the assumed constraints,

$$\left.\begin{aligned} P_{DY} &= 2.92(V/80.5)^3 + 1.20(V/86.5)^{1.7} \\ &\quad + 2.44(V/80.5) \\ P_{DY} &= 3.91(V/50)^3 + 1.63(V/50)^{1.7} \\ &\quad + 3.27(V/50) \end{aligned}\right\} \quad \text{(B7a)}$$

Equation (B7a) is also plotted in Fig. 12.

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5. Ronzi, R. C. and Rahnke, C. J., "Worldwide Automotive Powertrain Directions," SAE Paper 811377, Nov. 1981.

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TABLE I. - VEHICLE PARAMETER VALUES

Parameter	Vehicle		
	Baseline spark ignition	Baseline diesel	Improved aerody- namics
Mass kg, lb Economy test Performance test Inertia	 896 (1976) 987 (2176) 964 (2125)	 1016 (2240) 1107 (2440) 1077 (2375)	 896 (1976) 987 (2176) 964 (2125)
Dynamometer power absorber unit (PAU) setting, kW (hp)	5.2 (7.0)	5.1 (6.8)	4.1* (5.5)
Rolling resistance coefficient, C_R, N/kg (lb/lbm)	0.0981 (0.01)		
Drive axle	Front		
Mass on drive axle, percent	62		
Tire size, rev/km (rev/mi)	570 (917)		
Wheel base, cm (in)	230 (90.5)		
Movement of inertia for four wheel assemblies, kg-m^2 (lb-ft^2)	2.98 (70.6)		
Vehicle accessory	Alternator (see Fig. 1)		

*See Appendix B.

TABLE II. - PARAMETER VALUES FOR POWERTRAINS

Parameter	Transmission type		
	CVT	4 Speed manual	5 Speed manual
Transmission ratios	2.3 to 0.59	3.64 1.93 1.14 .86	3.17 1.94 1.29 .97 .76
Final drive ratio	6.24	3.47	3.90
Drive axle efficiency	Included in transmission efficiency	0.97	0.97
Additional gear losses, percent	4*	----	----
Engine accessories	See* Fig. 1	Included in engine maps	
Idle power, kW (hp)	0.8 (1.1)*	0.2 (0.3)	0.2 (0.3)

*With AGT engine.

TABLE III. - BASELINE VEHICLE FUEL ECONOMY RESULTS

(a) Spark-ignition engine; four-speed manual transmission; vehicle inertia mass, 964 kg (2125 lb)

Parameter	Measurement	Simulations			
Case identification	EPA test March 1981	Model validation	Added effect of:		
			C_D reduction to 0.39	Power scaling to 37 kW	Change to CVT
Fuel economy, km/l (mpg)					
Combined	17 (41)	17 (40)	18 (42)	19 (45)	20 (48)
City	15 (36)	15 (36)	16 (37)	17 (40)	18 (41)
Highway	20 (48)	20 (46)	21 (50)	22 (53)	25 (59)

(b) Diesel engine; five-speed manual transmission; vehicle inertia mass, 1077 kg (2375 lb)

Parameter	Measurement	Simulations				
Case identification	EPA test March 1980	Model validation	Added effect of:			
			C_D reduction to 0.39	Mass reduction to 964 kg	Power scaling to 37 kW	Change to CVT
Fuel economy, km/l (mpg)						
Combined	20 (47)	20 (46)	21 (50)	22 (52)	22 (51)	24 (57)
City	18 (42)	18 (42)	19 (45)	20 (47)	20 (46)	22 (52)
Highway	24 (56)	22 (53)	24 (58)	25 (60)	25 (59)	28 (66)

TABLE IV. - BASELINE VEHICLE WIDE-OPEN-THROTTLE ACCELERATION RESULTS

(a) Spark-ignition engine; four-speed manual transmission; vehicle inertia mass, 964 kg (2125 lb); kick-off speed, 750 rpm

Parameter	Simulation		
Case identification	Base vehicle	Added effect of:	
		Power scaling to 37 kW	Change to CVT
Time (0-97 kph; 0-60 mph), sec	15.9	19.4	16.5
Distance (2 sec), m (ft)	5.8 (19)	5.0 (16)	7.5 (25)
Distance (4 sec), m (ft)	24 (79)	20 (67)	28 (92)

(b) Diesel engine; five-speed manual transmission; vehicle inertia mass, 1077 kg (2375 lb); kickoff speed, 800 rpm

Parameter	Simulation			
Case identification	Base vehicle	Added effect of:		
		Mass reduction to 964 kg	Power scaling to 37 kW	Change to CVT
Time (0-97 kph; 0-60 mph), sec	24.4	21.4	20.0	16.3
Distance (2 sec), m (ft)	3.9 (13)	4.3 (14)	4.6 (15)	7.5 (25)
Distance (4 sec), m (ft)	15 (50)	17 (56)	18 (60)	28 (92)

TABLE V. - AGT DESIGN PARAMETER VALUES

Parameter	Values	
Power, kW (hp)	37 (50)	97 (130)
Peak turbine-inlet temperature, °C (°F)	1371 (2500)	
Peak regenerator-inlet temperature, °C (°F)	1093 (2000)	
Peak turbine-tip speed, m/s (ft/sec)	701 (2300)	
Ambient temperature, °C (°F)	29 (85)	
Compressor pressure ratio	5.0	
Compressor specific speed	0.753	
Shaft speed, rpm	156,000	100,000
Compressor efficiency	0.792	0.805
Turbine efficiency	0.836	0.854
Regenerator effectiveness	0.929	
Regenerator seal leakage, percent	6.3	4.1
Other flow leaks, percent	+20	As specified
Heat leaks, percent	+20	As specified
Relative component pressure drops	As specified	
Shaft losses, kW (hp)	1.2 (1.6)	2.1 (2.8)
Inlet mass-flow rate, kg/s (lb/sec)	0.16 (0.36)	0.39 (0.85)

TABLE VI. - EFFECT OF COMPRESSOR VARIABLE INLET GUIDE VANES (VIGV) ON FUEL ECONOMY. DIESEL FUEL; SEA LEVEL ON A 15° C (59° F) DAY; ENGINE IDLE SPEED, 60 PERCENT

Case	Fuel economy, km/l (mpg)		
	City	Highway	Combined
With VIGV	26 (61)	36 (84)	29 (69)
Without VIGV	22 (52)	34 (80)	26 (62)

TABLE VII. - COMPARISON OF FUEL ECONOMIES AMONG ENGINE TYPES. METAL-BELT CVT; VEHICLE INERTIA MASS, 964 kg (2125 lb); ENGINE RATED POWER, 37 kW (50 hp), VEHICLE C_D, 0.39

Engine type	Fuel economy, km/l (mpg)					
	Gasoline			Diesel fuel		
	City	Highway	Combined	City	Highway	Combined
Spark ignition	18 (41)	25 (59)	20 (48)	-------	-------	-------
Diesel	-------	-------	-------	22 (52)	28 (66)	24 (57)
Advanced gas turbine*	22 (53)	31 (73)	26 (60)	26 (61)	36 (84)	29 (69)

*60 percent idle speed; 15° C (59° F) at sea level.

TABLE VIII. - COMPARISON OF FUEL ECONOMIES AMONG ENGINE/TRANSMISSION TYPES.

INERTIA MASS, 964 kg (2125 lb); ENGINE RATED POWER,

37 kW (50 hp); VEHICLE C_D, 0.39

Engine/ transmission type	Fuel economy, km/l (mpg)					
	Gasoline			Diesel fuel		
	City	Highway	Combined	City	Highway	Combined
Spark ignition four-speed manual	17 (40)	22 (53)	19 (45)	-------	-------	-------
Diesel/five-speed manual	-------	-------	-------	20 (46)	25 (59)	22 (51)
Advanced gas turbine*/CVT	22 (53)	31 (73)	26 (60)	26 (61)	36 (84)	29 (69)

*60 percent idle speed; 15° C (59° F) at sea level.

TABLE IX. - SUMMARY AND COMPARISON OF WIDE-OPEN-THROTTLE

ACCELERATIONS. VEHICLE INERTIA MASS, 964 kg (2125 lb)

(a) Intermittant-combustion engines with manual transmissions

Parameter	Simulations		
Case identification	AGT/CVT*	Spark ignition 4 speed manual	Diesel 5 speed manual
Time (0-97 kph; 0-60 mph), sec	17.0	19.4	20.0
Distance (2-sec), m (ft)	5.0 (16)	5.0 (16)	4.6 (15)
Distance (4-sec), m (ft)	21 (70)	20 (67)	18 (60)

(b) All Engines with CVT

Parameter	Simulations		
Case identification	AGT*	Spark ignition	Diesel
Time (0-97 kph; 0-60 mph), sec	17.0	16.5	16.3
Distance (2-sec), m (ft)	5.0 (16)	7.5 (25)	7.5 (25)
Distance (4-sec), m (ft)	21 (70)	28 (92)	28 (92)

*Idle speed, 60 percent.

TABLE X. - TRANSMISSION EFFICIENCY COMPARISONS; AVERAGES OVER DRIVING CYCLES. VEHICLE INERTIA MASS, 964 kg (2125 lb); ENGINE RATED POWER, 37 kW (50 hp); VEHICLE C_D, 0.39

Engine/ transmission type	Average transmission efficiency		
	City	Highway	Combined
Spark-ignition/4 SM	0.83	0.88	0.85
Diesel/5 SM	.83	.89	.86
Spark-ignition/CVT	.84	.89	.86
Diesel/CVT	.84	.88	.86
AGT/CVT	.77	.82	.79

TABLE XI. - AGT VEHICLE FUEL ECONOMY SENSITIVITIES

Parameter	Range studied	Influence coefficients	Slope
Vehicle inertia mass, kg (lb)	873 to 1055	-0.37	$-0.011 \frac{km/l}{kg}$
	(1925 to 2325)		$-0.012 \frac{mpg}{lb}$
Additional gear efficiency	0.90 to 1.00	0.66	$0.20 \frac{km/l}{pt}$
			$0.48 \frac{mpg}{pt}$
Regenerator effectiveness	0.909 to 0.949	1.49	$0.45 \frac{km/l}{pt}$
			$1.07 \frac{mpg}{pt}$
Turbine efficiency	0.818 to 0.854	1.08	$0.32 \frac{km/l}{pt}$
			$0.76 \frac{mpg}{pt}$
Compressor efficiency	0.773 to 0.812	0.76	$0.26 \frac{km/l}{pt}$
			$0.62 \frac{mpg}{pt}$
Regenerator seal leakage, percent	2 to 8	-0.05	$-0.22 \frac{km/l}{\%}$
			$-0.52 \frac{mpg}{\%}$
Other flow leaks, multiple of reference	1 to 2	-0.03	
Heat leaks, multiple of reference	1 to 2	-0.03	
Idle power, kW (hp)	2 to 2.2	-0.03	$-1.00 \frac{km/l}{kW}$
	(0 to 3.0)		$-1.75 \frac{mpg}{hp}$
Shaft losses, kW (hp)	2 to 2.2	-0.11	$-2.64 \frac{km/l}{kW}$
	(0 to 3.0)		$-4.64 \frac{mpg}{hp}$

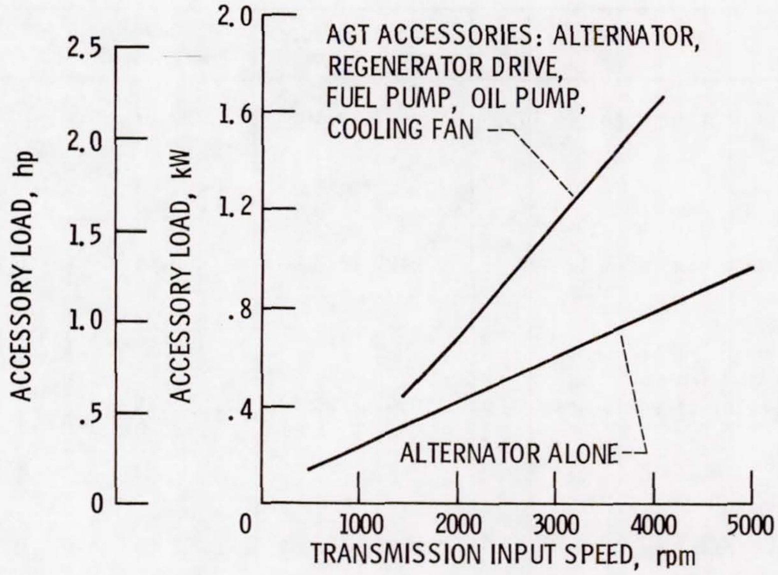

Figure 1. - Vehicle and AGT accessory load requirements.

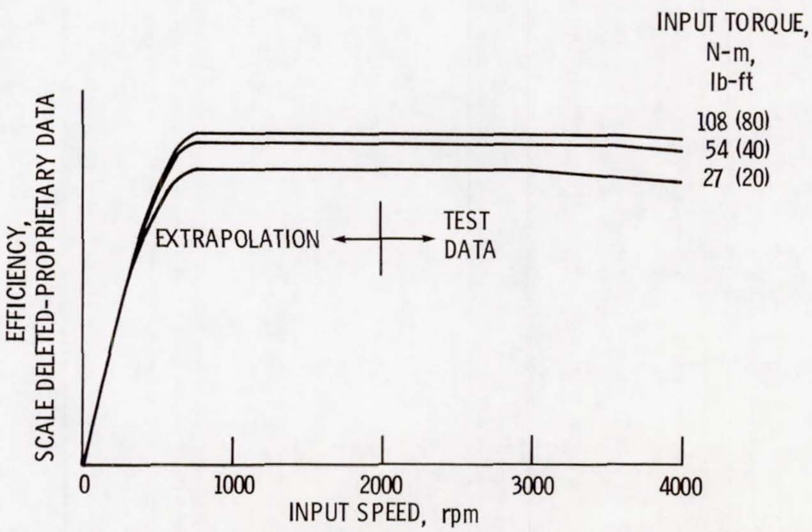

Figure 2. - Typical CVT test data form and extrapolation. Transmission belt ratio, 2.30.

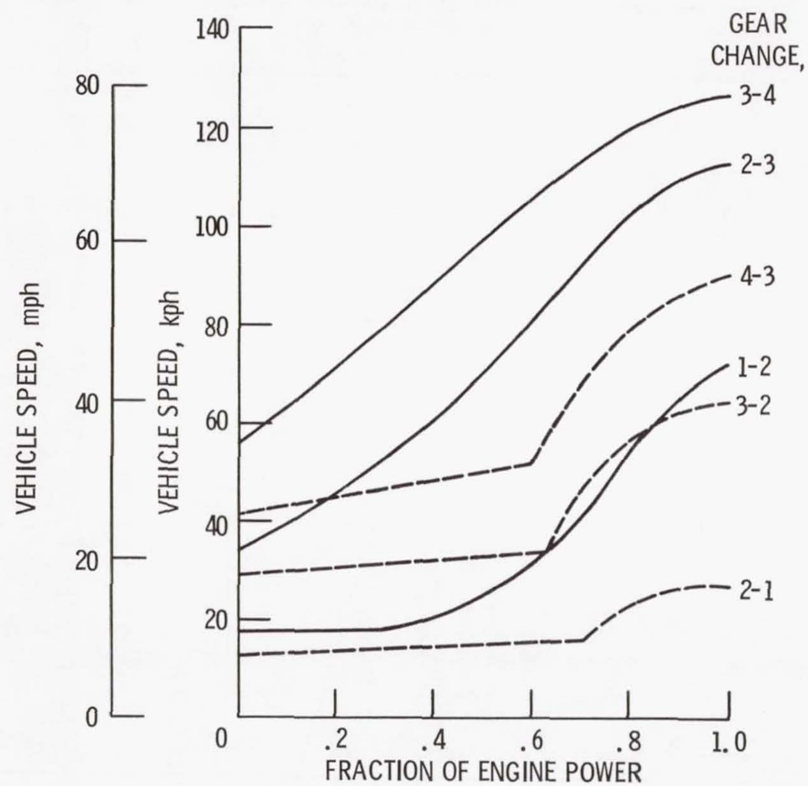

Figure 3. - Typical shift schedule. Four-speed manual transmission.

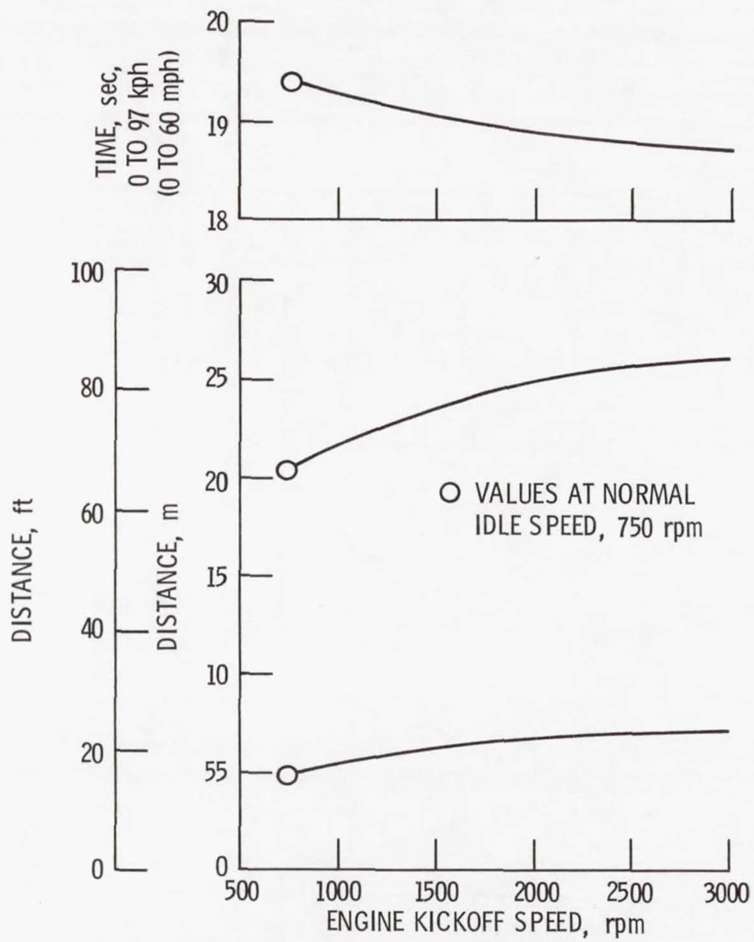

Figure 4. - Effect of kickoff speed on vehicle acceleration. Spark-ignition engine, four-speed manual transmission; vehicle inertia mass, 964 kg (2125 lb); engine rated power, 37 kW (50 hp); vehicle C_D, 0.39.

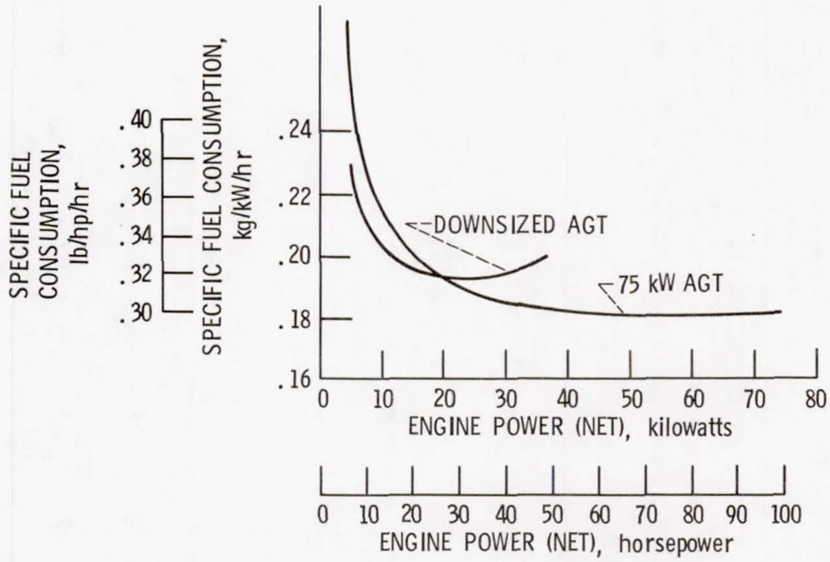

Figure 5. - Specific fuel consumption comparison between AGT engines. VIGV settings optimized; 15° C (59° F); sea level.

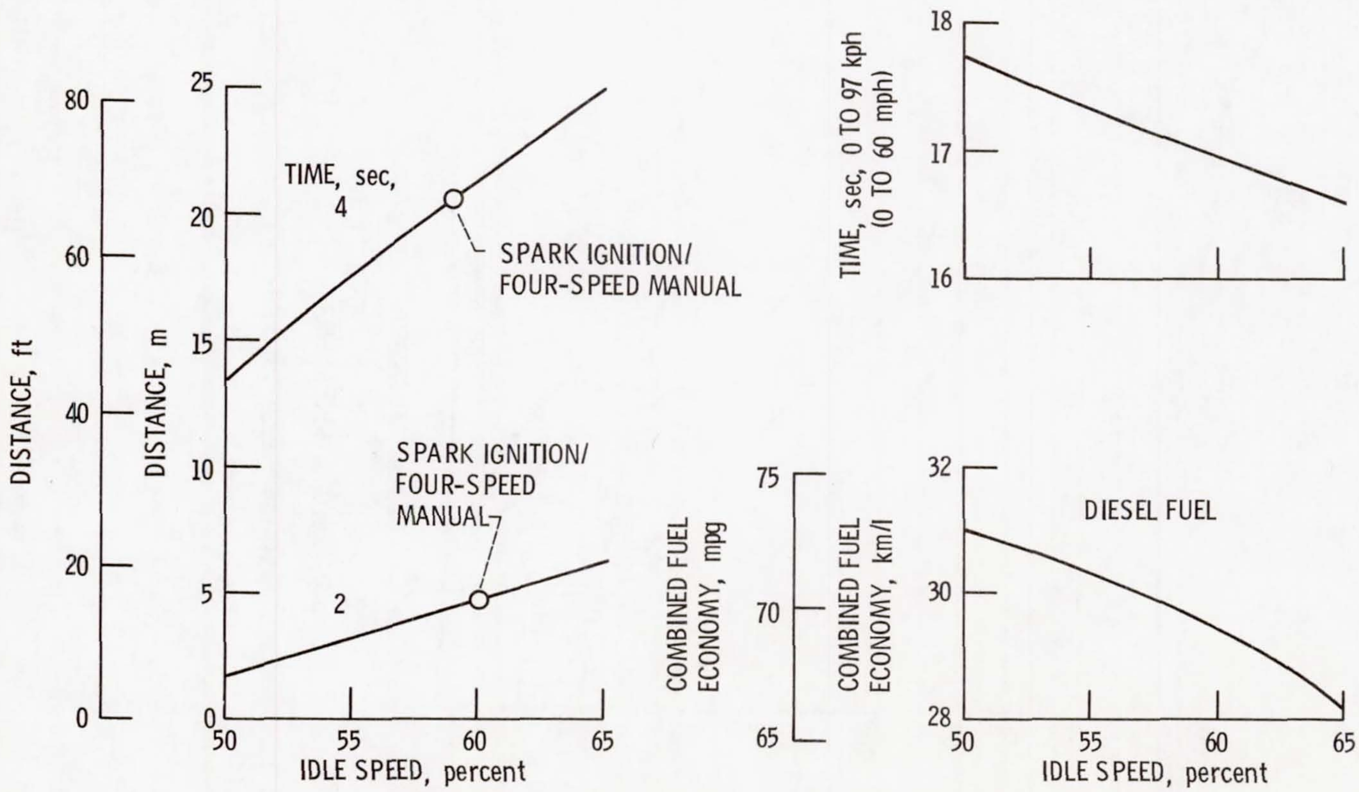

Figure 6. - Trade offs and selection of AGT idle speed. Sea level on a 15° C (59° F) dag.

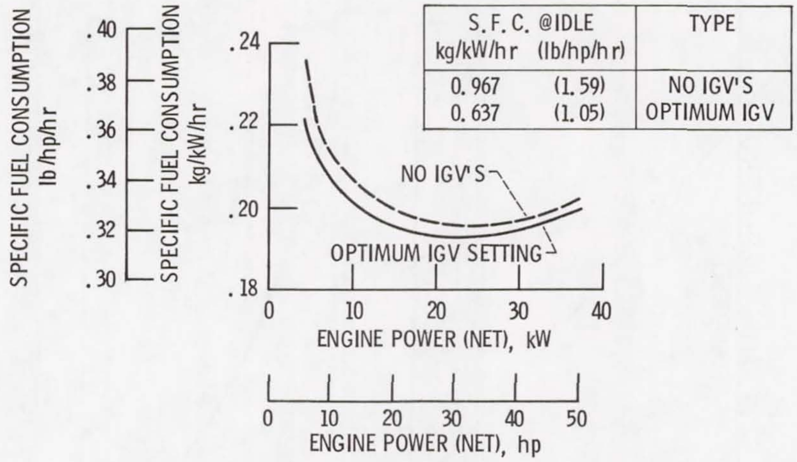

Figure 7. - Effect of variable compressor-inlet guide vanes (IGV) on specific fuel consumption. Diesel fuel; 15° C (59° F); sea level; engine idle speed, 60%.

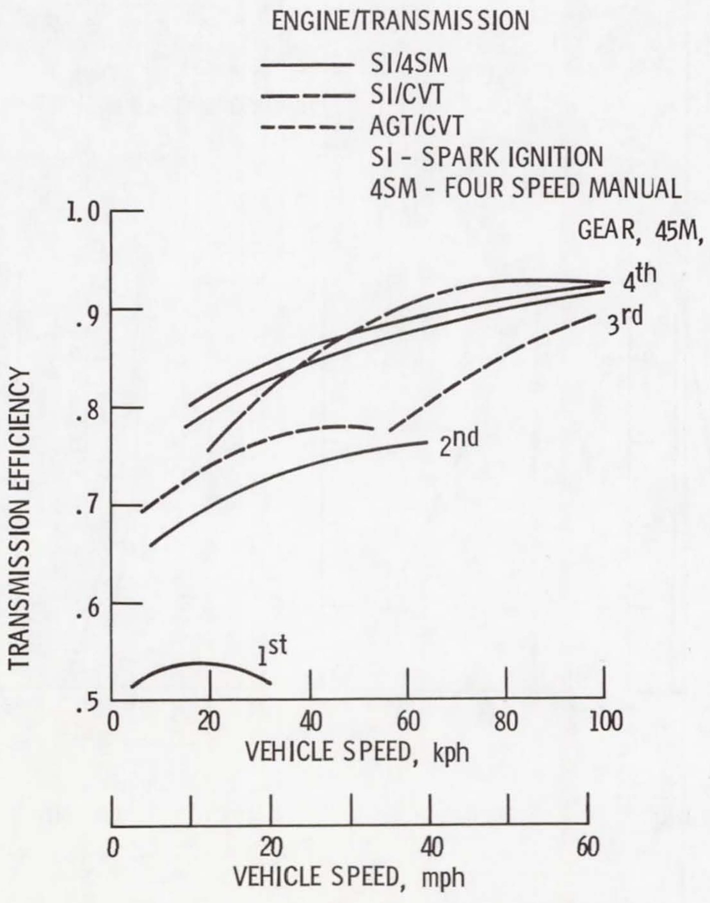

Figure 8. - Transmission efficiency comparisons: constant vehicle speed. Vehicle inertia mass, 964 kg (2125 lb); engine rated power, 37 kW (50 hp); vehicle C_D, 0.39.

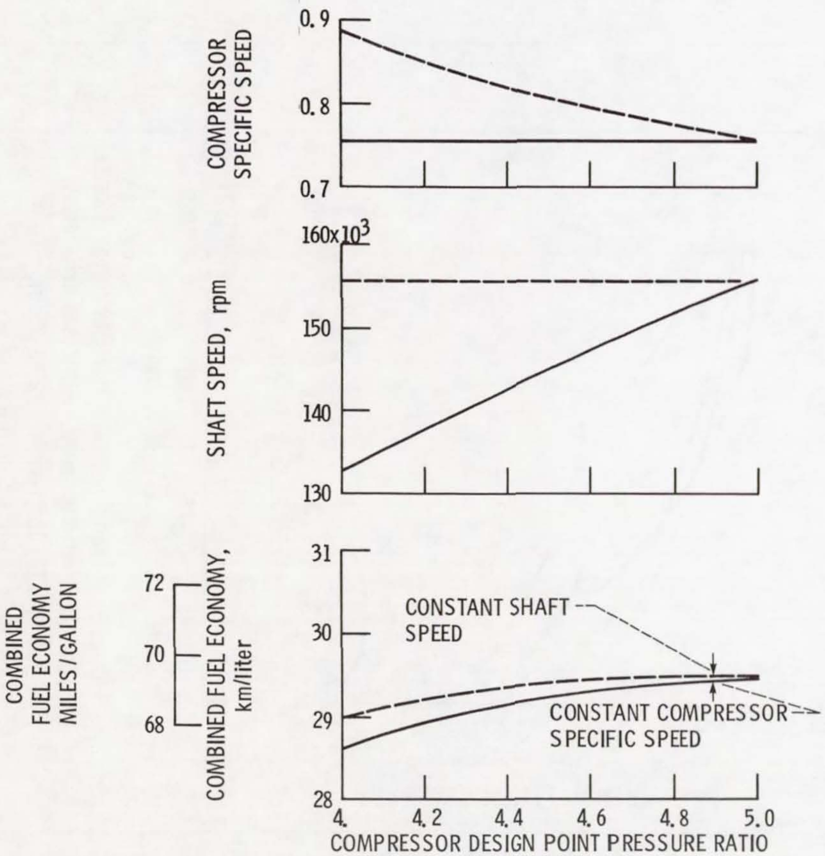

Figure 9. - Effect of compressor design pressure ratio on vehicle combined fuel economy.

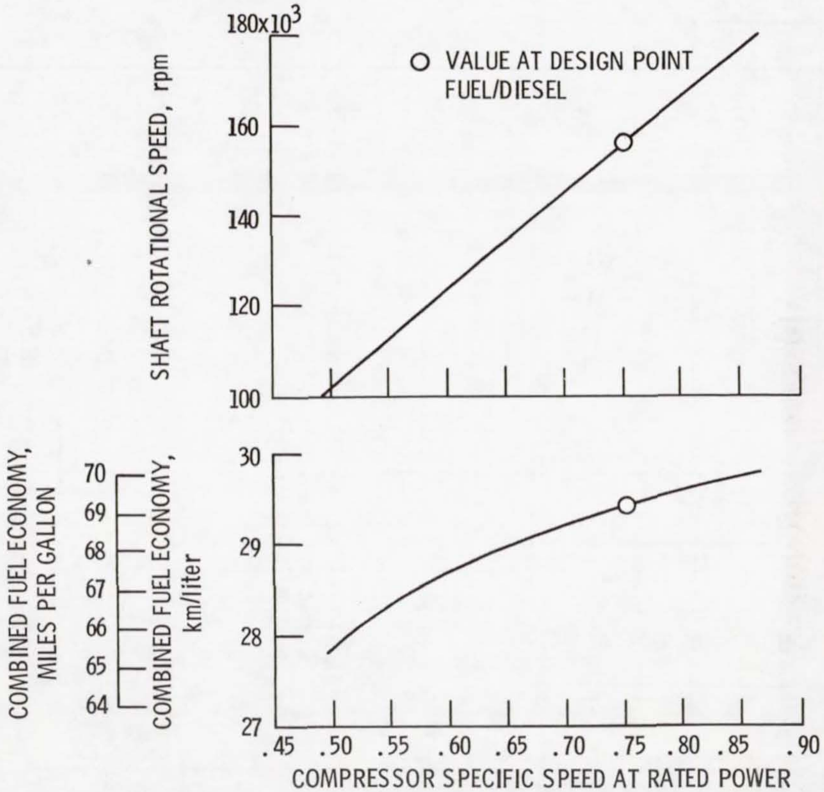

Figure 10. - Effect of compressor specific speed at rated power on vehicle combined fuel economy. Compressor pressure ratio constant (P_r = 5).

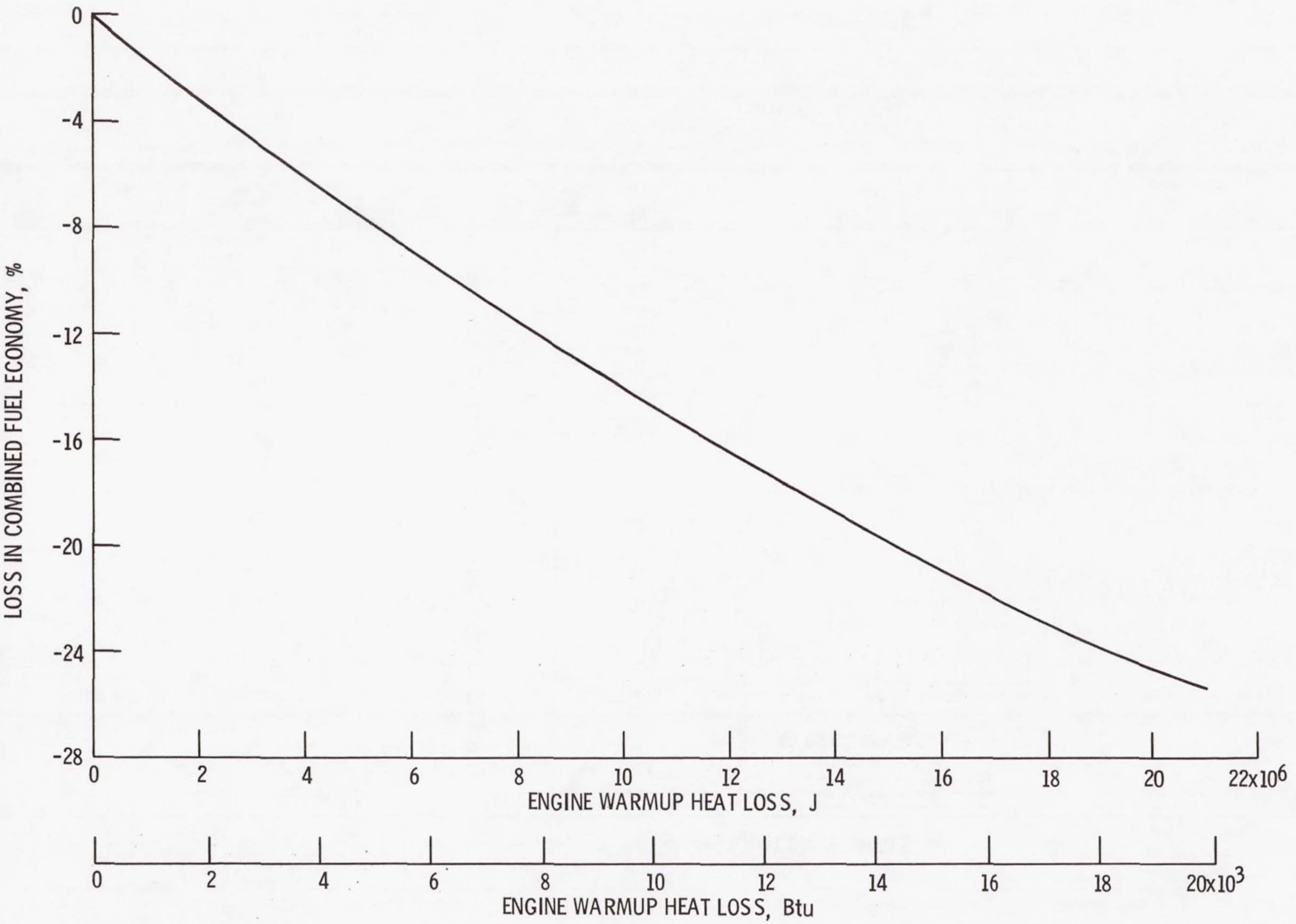

Figure 11. - Sensitivity of downsized AGT engine combined fuel economy to warm up heat loss. Assumed CVT transmission, and $C_D = 0.39$.

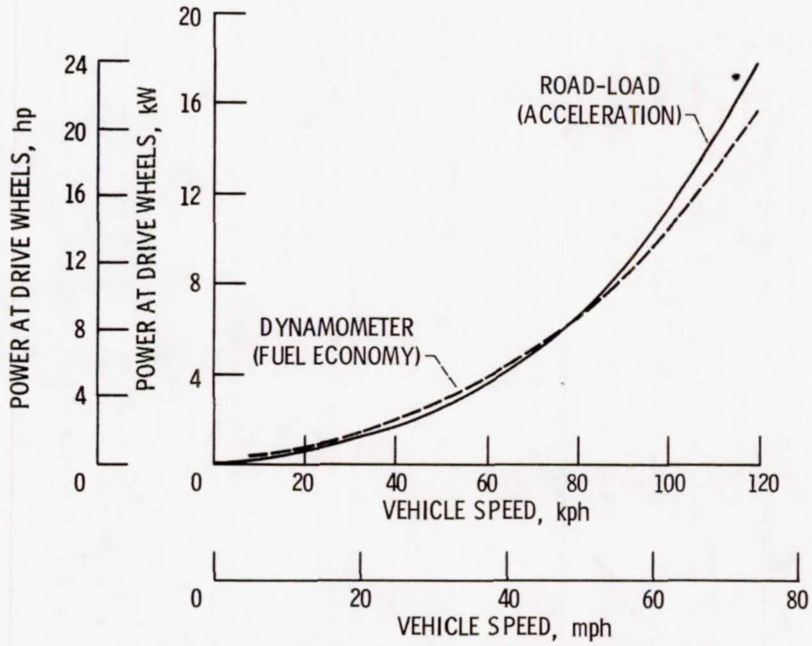

Figure 12. - Comparison of road-load and dynamometer power requirements.

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16. Abstract

A study was conducted to see if relative fuel economy advantages exist for a ceramic turbine engine when it is downsized for a small car. A 75 kW (100 hp) single-shaft engine currently under development was analytically downsized to 37 kW (50 hp) and analyzed with a metal-belt continuously variable transmission in a synthesized car. With gasoline, a 25-percent advantage was calculated over that of a current spark-ignition engine, scaled to the same power, using the same transmission and car. With diesel fuel, a 21-percent advantage was calculated over that of a similar diesel-engine vehicle.

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